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Theoretical And Experimental Analysis Of Supercritical Carbon Dioxide Cooling

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ABSTRACT

With on-going developments in the field of trans-critical carbon dioxide (R-744) vapour compression cycles, a need to effectively describe the heat transfer of supercritical carbon dioxide for application in larger diameter tube-in-tube heat exchangers was identified. This study focuses on the in-tube cooling of supercritical carbon dioxide for application in the gas cooler of a trans-critical heat pump. A literature study has revealed there are Nusselt number correlations specifically developed for the cooling of supercritical carbon dioxide. These correlations were proven to be accurate only for certain operating conditions and tube geometries. From the literature review a shortcoming was identified for a generic heat transfer correlation that can be applied over a wide range of fluid conditions for supercritical carbon dioxide cooling. The objective of this study was to compare experimental data obtained from a trans-critical heat pump test bench with different Nusselt number correlations available in literature. The experimental tube diameter used for this study (16mm), which was based on commercially available pipe sizes, was considerably larger than the validated tube diameters used by the researchers who developed Nusselt number correlations specifically for the supercritical cooling of carbon dioxide. The experimental Reynolds number ranges were very high ($350,000 < Re < 680,000$) compared to those found in literature ($Re < 300,000$). Experimental results showed that correlations specifically developed for supercritical carbon dioxide cooling generally over-predicts the Nusselt numbers with an average relative error of between 62% and 458% and subsequently also over-predicts the convection heat transfer coefficients. Furthermore, generic heat transfer correlations were compared to the experimental results which over-predicted the Nusselt number with an average relative error of between 20% and 45% over the entire Reynolds number range. More specifically, the correlation by Dittus and Boelter (1985) correlated with an average relative error of 9% for $350,000 < Re < 550,000$. From the results of this study it was concluded that cooling heat transfer of supercritical carbon dioxide in larger tube diameters and at higher Reynolds numbers is more accurately predicted by the generic Dittus and Boelter (1985) and Gnielinski (1975) correlations mainly due to the absence of thermo-physical property ratios as seen in the CO_2 -specific correlations.

1. INTRODUCTION

In the Heating, Ventilation, Air Conditioning and Refrigeration (HVAC&R) industry, working fluids (or refrigerants) are continuously changing due to the ever growing demand of the industry. Over the last decade, the trend in the industry was towards refrigerants that has a low Global Warming Potential (GWP) and are not ozone depleting. Most synthetic refrigerants are fluorine - and/or chlorine based substances that have a negative impact on climate change (Kim *et al.*, 2004). Thus carbon dioxide has gained renewed interest as a refrigerant in recent years. Being a non-toxic, inexpensive, natural gas that has a zero net impact on global warming, it is readily accepted as a good alternative by many governments and environmentalists (Calm, 2008). Carbon dioxide (CO_2), or R-744 as it is known in the HVAC&R industry, has certain interesting properties that make it very unique among refrigerants. The low critical temperature of 31.1 °C at a high pressure of 7.29MPa poses challenges when used in a vapour compression cycle (Kim *et al.*, 2004). Heat pump cycles using carbon dioxide as refrigerant must operate in a trans-

critical fashion to be efficient and competitive (Austin and Sumathy, 2011). Heat rejection takes place in a gas cooler operating at very high gas temperatures and pressures as found in a trans-critical carbon dioxide cycle.

In order to successfully simulate or design a trans-critical heat pump with carbon dioxide as refrigerant, accurate Nusselt number (Nu) correlations describing the heat transfer must be acquired. Studies have been conducted to measure experimental Nu values and compare them with theoretical Nu values in order to predict the heat transfer when cooling supercritical carbon dioxide (Cheng *et al.*, 2008). The main shortcomings from these correlations are the narrow operating conditions and geometries to which they can be applied. These correlations can be classified into two main geometry classes, namely micro-tubes ($D_h < 3\text{mm}$) and macro-tubes ($D_h > 3\text{mm}$). Table 1 summarizes the experimental studies that were done on cooling of supercritical CO_2 for application in a vapour compression cycle. From these studies, there appears to be no generalised correlation to describe heat transfer. Only six of these studies (Yoon *et al.*, 2003, Pitla *et al.*, 2001, Dang and Hihara, 2004, Son and Park 2006, Oh and Son, 2010 and Zhao and Jiang, 2011) considered macro-scale channel geometry for their experiments. All six of these studies came forth with new heat transfer correlations for super critical cooling of carbon dioxide, but based on the commonly used Nu correlations of either Gnielinski (1975) or Dittus and Boelter (1985). These studies considered tube diameters between 3mm and 7.75mm.

Table 1: A summary of experimental studies done on cooling of supercritical CO_2 (Cheng *et al.*, 2008).

Author	Tube diameter (mm)	Inlet Temperature range ($^{\circ}\text{C}$)	Inlet Pressure range (MPa)
Yoon <i>et al.</i> (2003)	7.73	50-80	7.5-8.8
Pettersen (2000)	0.79	15-70	8.1-10.1
Pitla <i>et al.</i> (2001)	4.72	120	8 - 12
Dang and Hihara (2004)	1-6	30-70	8-10
Mori <i>et al.</i> (2003)	6	20-70	9.5
Huai <i>et al.</i> (2005)	1.31	22-53	7.4-8.5
Son and Park (2006)	7.75	90-100	7.5-10
Dang <i>et al.</i> (2007)	1,2,4,6	20-70	8-10
Oh and Son (2010)	4.55, 7.75	90-100	90-100
Zhao and Jiang (2011)	4.01	80-140	4.5-5.5

The experimental heat pump test facility at the North-West University (NWU) utilizes commercially available tubing with an inner diameter of 16mm in a tube-in-tube configuration for the gas cooling process. The absence of proven Nu correlations to describe heat transfer for a system with larger geometry tubing creates an opportunity for contribution to the existing field of knowledge. The objective of the study was to analyse the cooling of supercritical carbon dioxide by applying eight different Nu correlations for heat transfer in an analysis model and compare the results with experimental data obtained from the test facility. Six of the eight correlations tested were developed specifically for supercritical cooling of CO_2 . The remaining two correlations were the correlations of Gnielinski (1975) and Dittus and Boelter (1985).

2. EXPERIMENTAL APPARATUS AND PROCEDURE

2.1 Test facility

A trans-critical carbon dioxide heat pump forms the basic layout of the NWU test facility. As shown in Figure 1, the test facility consists of four main components namely the compressor, CO_2 -to-water gas cooler, expansion valve and water-to- CO_2 evaporator. The test facility is further equipped with: i) mass flow meters on the water and gas sides, ii) temperature transmitters on the water and gas sides, iii) pressure transmitters on the gas side, iv) Variable Frequency Drive (VFD) on the compressor, v) motorized automation on the expansion valve, vi) thermal insulation on all piping and heat exchangers, vii) data logging equipment, and viii) water pumps, flow switches and manual valves to control water flows. A tube-in-tube configuration was used for the heat exchangers with carbon dioxide flowing in a 16mm inner diameter, schedule 40 AISI 304 stainless steel tube, with the water in the annular section as seen in Figure 2.

The compressor is a semi hermetic, reciprocating type specifically designed for carbon dioxide. The maximum pressure on the outlet is rated at 13.0MPa, with a fluid displacement of 9.2 m³/h. The VFD regulates the electrical supply to the compressor for capacity control. The mass flux is varied by controlling the electrical phase frequency between 30 – 70 Hz.

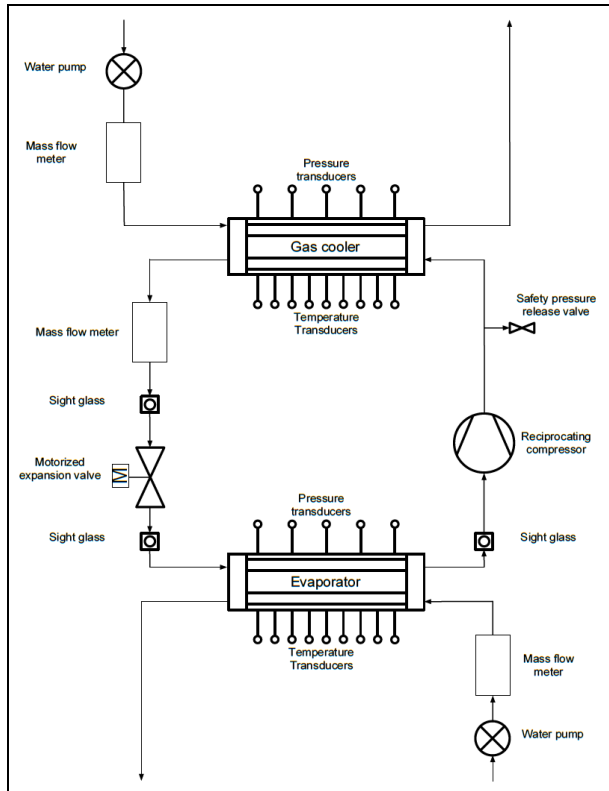


Figure 1: Test bench layout.

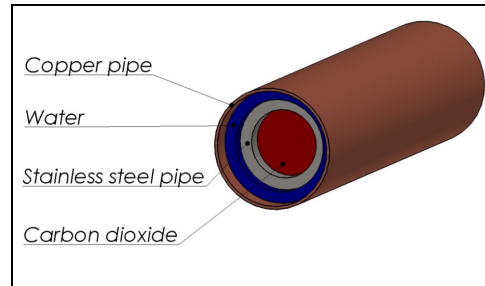


Figure 2: Heat exchanger flow channels.

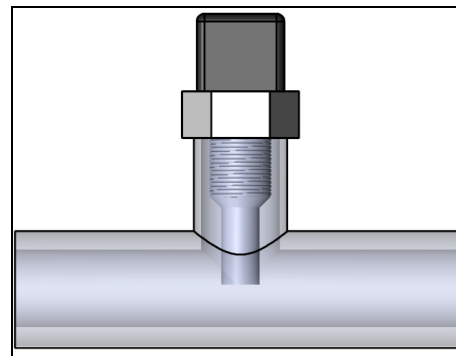


Figure 3: Temperature transmitter shown in the mounted position with a Pt1000 element in direct contact with the bulk fluid.

The test facility was fitted with instrumentation to evaluate real time data and store it for further processing. Six pressure transmitters were installed at preselected positions (Figure 1) on the gas cooler to measure the CO_2 pressure. The accuracy is rated at $\pm 0.3\%$ of the measured value (including linearity, hysteresis and repeatability analysis). Temperature transmitters were installed in two meter increments on both heat exchangers to measure the bulk fluid temperatures. A thermal pocket brings the Pt1000 element in contact with the bulk fluid in the centre of the tube (Figure 3). The accuracy is rated at a tolerance of t and can be calculated by equation (1).

$$t = \pm (0.3 + 0.005 \times T) \quad (1)$$

The CO_2 mass flow is measured by a coriolis mass flow meter and water flow is measured on both the gas cooler and the evaporator by magnetic mass flow meters. The accuracy for flow measurements is rated at $\pm 0.5\%$ of the reading. The combined uncertainty associated with the instrumentation resulted in an average uncertainty of 0.18% for Q_{water} values and 0.81% for Q_{CO2} values.

2.2 Data processing procedure

It is required to calculate values for Nu_{theo} employing various Nu correlations for the same fluid conditions as observed in the experiments from which Nu_{exp} are determined. The values for Nu_{theo} are calculated by a procedure illustrated in Figure 4. The inlet and outlet parameters are fixed to the experimental values for the increment under evaluation. The averaged fluid conditions per increment is calculated and used to determine fluid properties for each increment. Component characteristics can be calculated once the fluid properties are known, including Reynolds number, Prandtl number and the friction factor. The different Nu number correlations are then employed to calculate

Nu_{theo} values under the same conditions as experimentally measured. The Nu correlations employed in this study are summarised in equations (2) to (10) below:

Correlation by Dittus and Boelter (1985):

$$Nu_{DB} = 0.023 Re_b^{0.8} Pr_b^n$$

$$n = \begin{cases} 0.4 & \text{if } \frac{T_w}{T_b} > 1 \\ 0.3 & \text{if } \frac{T_w}{T_b} < 1 \end{cases} \quad (2)$$

Correlation by Gnielinski (1975):

$$Nu_G = \frac{f_f}{8} (Re_b - 1000) Pr_b \left[1.07 + 12.7 \sqrt{\frac{f_f}{8}} (Pr_b^{2/3} - 1) \right]^{-1} \quad (3)$$

Correlation by Pitla et.al. (2001):

$$Nu_p = \left(\frac{Nu_w + Nu_b}{2} \right) \frac{k_w}{k_b} \quad (4)$$

where Nu_w and Nu_b are Nusselt numbers calculated with the Gnielinski (1975) correlation (3) using thermo-physical properties evaluated at the bulk temperature and wall temperature respectively.

Correlation by Yoon et.al. (2003):

$$Nu_Y = 0.14 Re_b^{0.69} Pr_b^{0.66} \quad \text{for } T_b > T_{pc}$$

$$Nu_Y = 0.013 Re_b Pr_b^{-0.05} \left(\frac{\rho_{pc}}{\rho_b} \right)^{1.6} \quad \text{for } T_b \leq T_{pc} \quad (5)$$

Correlation by Dang and Hihara (2004):

$$Nu_{D\&H} = \frac{f_{f,f}}{8} (Re_b - 1000) Pr \left[1.07 + 12.7 \sqrt{\frac{f_{f,f}}{8}} (Pr^{2/3} - 1) \right]^{-1}$$

$$Pr = \begin{cases} c_{p_b} \mu_b / k_b, & \text{for } c_{p_b} > \tilde{c}_p \\ \tilde{c}_p \mu_b / k_b, & \text{for } c_{p_b} < \tilde{c}_p \text{ and } \mu_b / k_b \geq \mu_f / k_f \\ \tilde{c}_p \mu_f / k_f, & \text{for } c_{p_b} < \tilde{c}_p \text{ and } \mu_b / k_b < \mu_f / k_f \end{cases} \quad (6)$$

$$\tilde{c}_p = (h_b - h_w) / (T_b - T_w)$$

$$f_{f,f} = (1.82 \log Re_f - 1.64)^{-2}$$

where \tilde{c}_p denotes an integrated specific heat in the radial direction and $f_{f,f}$ the Filonenko friction factor evaluated at the film temperature (T_f) which is calculated by:

$$T_f = \frac{(T_b + T_w)}{2} \quad (7)$$

Correlation by Son and Park (2006):

$$\begin{aligned}
 Nu_{S\&P} &= Re_b^{0.55} Pr_b^{0.23} \left(\frac{c_{p_b}}{c_{p_w}} \right)^{0.15} & \text{for } \frac{T_b}{T_{pc}} > 1 \\
 Nu_{S\&P} &= Re_b^{0.35} Pr_b^{1.9} \left(\frac{\rho_b}{\rho_w} \right)^{-1.6} \left(\frac{c_{p_b}}{c_{p_w}} \right)^{-3.4} & \text{for } \frac{T_b}{T_{pc}} \leq 1
 \end{aligned} \tag{8}$$

Correlation by Oh and Son (2010):

$$\begin{aligned}
 Nu_{O\&S} &= 0.023 Re_b^{0.7} Pr_b^{2.5} \left(\frac{c_{p_b}}{c_{p_w}} \right)^{-3.5} & \text{for } \frac{T_b}{T_{pc}} > 1 \\
 Nu_{O\&S} &= 0.023 Re_b^{0.6} Pr_b^{3.2} \left(\frac{\rho_b}{\rho_w} \right)^{3.7} \left(\frac{c_{p_b}}{c_{p_w}} \right)^{-4.6} & \text{for } \frac{T_b}{T_{pc}} \leq 1
 \end{aligned} \tag{9}$$

Correlation of Zhao and Jiang (2011):

$$Nu_{Z\&J} = \frac{\frac{f_f}{8} (Re_b - 1000) Pr_b}{1.07 + 12.7 \sqrt{\frac{f_f}{8} (Pr_b^{2/3} - 1)}} \left[1 + \left(\frac{D_H}{L} \right)^{2/3} \right] C_{vp}$$

(10)

with

$$C_{vp} = \begin{cases} 0.93 \left(\frac{Pr_w}{Pr_b} \right)^{-0.11} \left(\frac{\bar{c}_p}{c_{p_b}} \right)^{0.96} \left(\frac{\rho_w}{\rho_b} \right)^{1.06} & \text{for } \frac{T_b}{T_{pc}} \leq 1 \\ 1.07 \left(\frac{T_w}{T_b} \right)^{-0.45} \left(\frac{\bar{c}_p}{c_{p_b}} \right)^{0.61} \left(\frac{\rho_w}{\rho_b} \right)^{-0.18} & \text{for } \frac{T_b}{T_{pc}} > 1 \end{cases}$$

$$\bar{c}_p = \frac{h_i - h_e}{T_i - T_e}$$

where C_{vp} denotes the property variation coefficient and \bar{c}_p the mean value of specific heat calculated over the entire length of the test section.

The Nu_{exp} must be calculated for each test section. The path of heat transfer from the hot CO_2 to the cold water involves convection on the inner surface of the thick-walled stainless steel pipe, conduction in the radial direction through the stainless steel wall and convection from the stainless steel wall to the water in the annular area. Conduction heat transfer in the axial direction of the fluid and stainless steel pipe was assumed to be negligible (Pitla *et al.*, 2001). The $CO_2 Nu_{exp}$ in each sub-section of the heat exchanger was calculated using the procedure illustrated in Figure 5. Using the conservation of energy, the heat transfer rate was calculated from the change in enthalpy in the CO_2 over a test section as shown by equation (11).

$$\dot{Q}_{CO_2} = \dot{m}_{CO_2} (h_{CO_2,out} - h_{CO_2,in}) \tag{11}$$

Heat transfer rate between two fluids can also be calculated using equation (12).

$$\dot{Q} = U \cdot A \cdot LMTD \tag{12}$$

The $LMTD$ value is calculated by equation (13).

$$LMTD = \frac{\Delta T_1 - \Delta T_0}{\ln \frac{\Delta T_1}{\Delta T_0}} \tag{13}$$

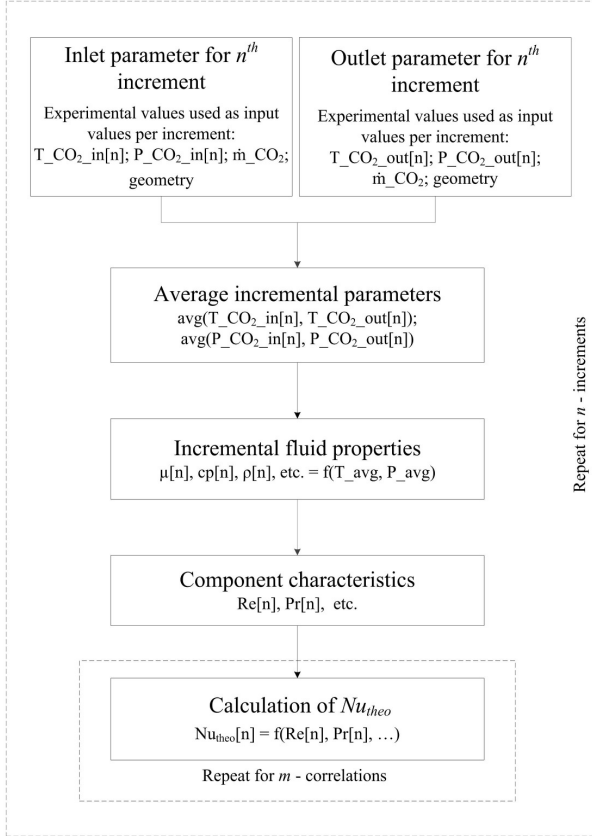


Figure 4: Calculation of Nu_{theo} - analysis program flow diagram.

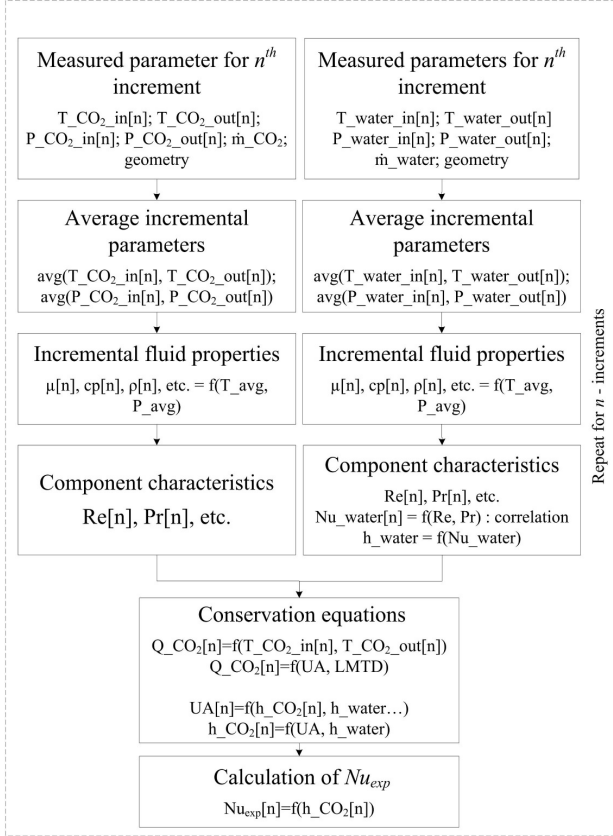


Figure 5: Calculation of Nu_{exp} - analysis program flow diagram.

with the temperature differences calculated by

$$\begin{aligned}\Delta T_1 &= T_{CO_2, out} - T_{water, out} \\ \Delta T_0 &= T_{CO_2, in} - T_{water, in}\end{aligned}\quad (14)$$

The overall heat transfer coefficient (U) in equation (15) is defined by the sum of three terms of heat transfer, namely: i) forced convective heat transfer from carbon dioxide in the inner tube, ii) conduction heat transfer through the stainless steel tube and iii) convective heat transfer to the water.

$$\frac{1}{U \cdot A} = \frac{1}{h_{c, CO_2} A_i} + \frac{\ln(D_o/D_i)}{2\pi kL} + \frac{1}{h_{c, water} A_o} \quad (15)$$

The \dot{Q}_{CO_2} value over each increment is used to calculate $U \cdot A$ from equation (12). The conduction term is calculated from the thermal conductivity of stainless steel and dimensions of the tube. The forced convection term on the water side can be calculated using a water heat transfer correlation. The correlation of Gnielinski (1975) ($3,000 \leq Re \leq 5 \times 10^6$) was chosen to accommodate Reynolds numbers ranging from 5,000 to 12,000 on the water side. The h_{c, CO_2} value can be calculated since all other variables in equation (15) are known. Finally the Nu_{CO_2} values can be calculated by relating h_{c, CO_2} with Nu_{CO_2} by using equation (16).

$$Nu_{CO_2} = \frac{h_{c, CO_2} D_H}{k_b} \quad (16)$$

3. RESULTS

3.1 Experimental uncertainty propagation analysis

For each temperature, pressure and mass flow data point captured, a total uncertainty value is calculated. The raw temperature data was smoothed using a regression model where the total standard uncertainty of the mathematical function was derived while incorporating the instrumentation and statistical uncertainties in the calculation. The total uncertainty band of the regressed temperature data is shown in Figures 6 and 7. The propagation of uncertainty from measured values to calculated Nu_{exp} numbers were calculated for each position (data point) in the gas cooler as shown in all Nu plots.

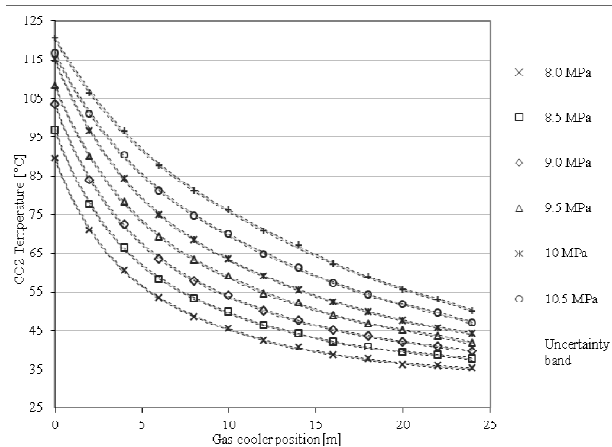


Figure 6: Experimental carbon dioxide temperatures along the gas cooler position.

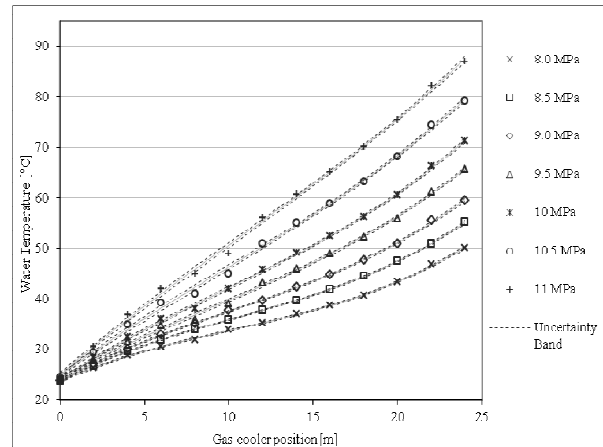


Figure 7: Experimental water temperatures along the gas cooler position.

3.2 Comparison of Nu_{theo} to Nu_{exp}

The platform of comparison used in this study was to plot Nu_{exp} as well as Nu_{theo} values against non-dimensional parameters such as the Re number or Pr number which not only provides a non-dimensional platform, but also reveals more about the type of flow at the specific point, as shown in Figure 8 and 9.

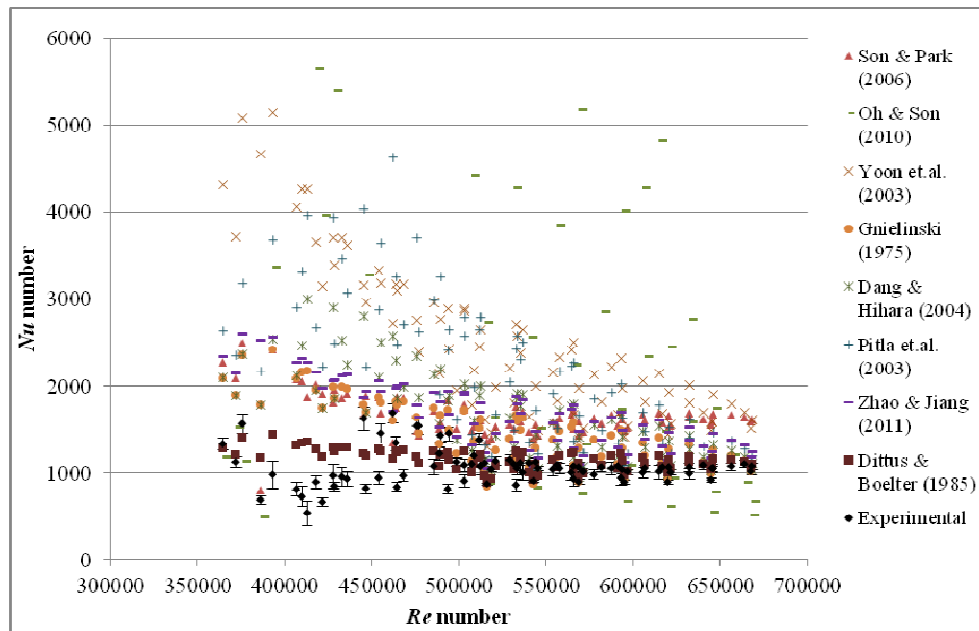


Figure 8: Nu numbers plotted against Re numbers. This plot combines all data sets (8MPa – 11MPa) by plotting against the non-dimensional Re number.

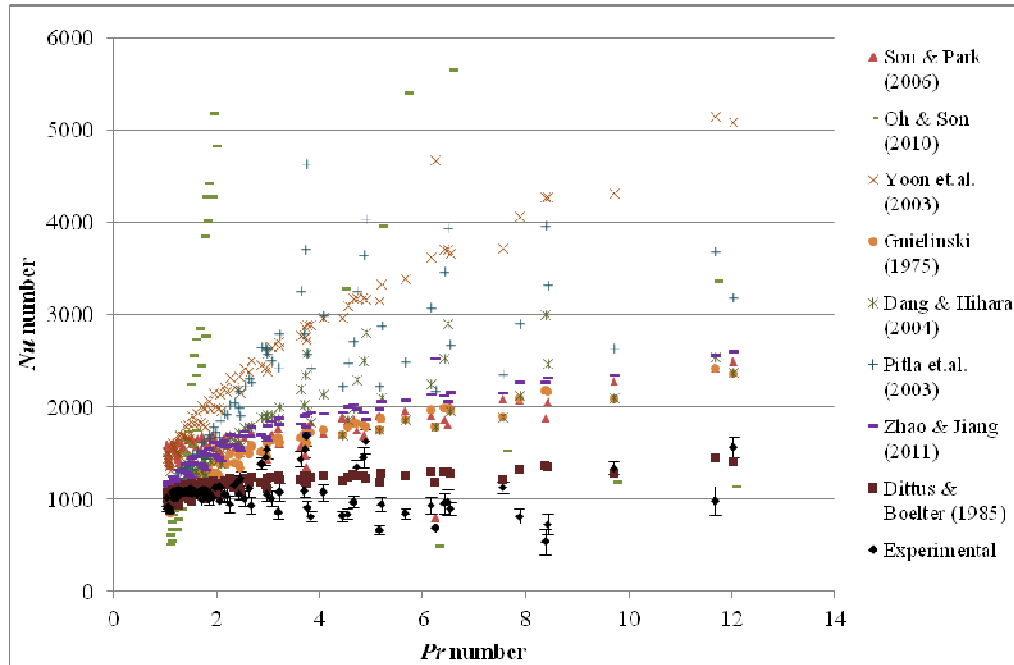


Figure 9: Nu numbers plotted against Pr numbers. This plot combines all data sets (8MPa – 11MPa) by plotting against the non-dimensional Pr number

Table 2: Summary of results obtained.

<u>Nu Correlation</u>	<u>Average relative error (%)</u>
Dittus and Boelter (1985)	20%
Son and Park (2006)	66%
Yoon <i>et al.</i> (2003)	143%
Oh and Son (2010)	458%
Gnielinski (1975)	45%
Pitla <i>et al.</i> (2002)	107%
Dang and Hihara (2004)	73%
Zhao and Jiang (2011)	62%

All datasets from different pressures are combined to get data over a wide spectrum of flow conditions. Figure 8 and Figure 9 show the experimental and theoretical data plotted against Re and Pr numbers. Each individual dataset follows a unique trend. With the Nu_{exp} remaining relatively constant, the majority of correlations over-predict Nu for all Re , but to a lesser extent as the Re number increases and similarly as Pr decreases. The average percentage error for each correlation is shown in Table 2.

3.3 Effect of pseudo critical point

Vast property changes occur when supercritical CO_2 approaches the pseudo-critical point. It is therefore important to consider the effect of the pseudo critical point in the temperature glide seen in a gas cooler and on the heat transfer and Nu correlations. For the pressures from 8 – 11 MPa, the pseudo critical temperatures range from 34.7 – 49.7 °C. The Nu_{exp} numbers plotted against T_{CO_2} are shown in Figure 10. It can be seen that the thermo-physical property variations near the pseudo critical point have a much larger effect on the correlated values than on the experimental data.

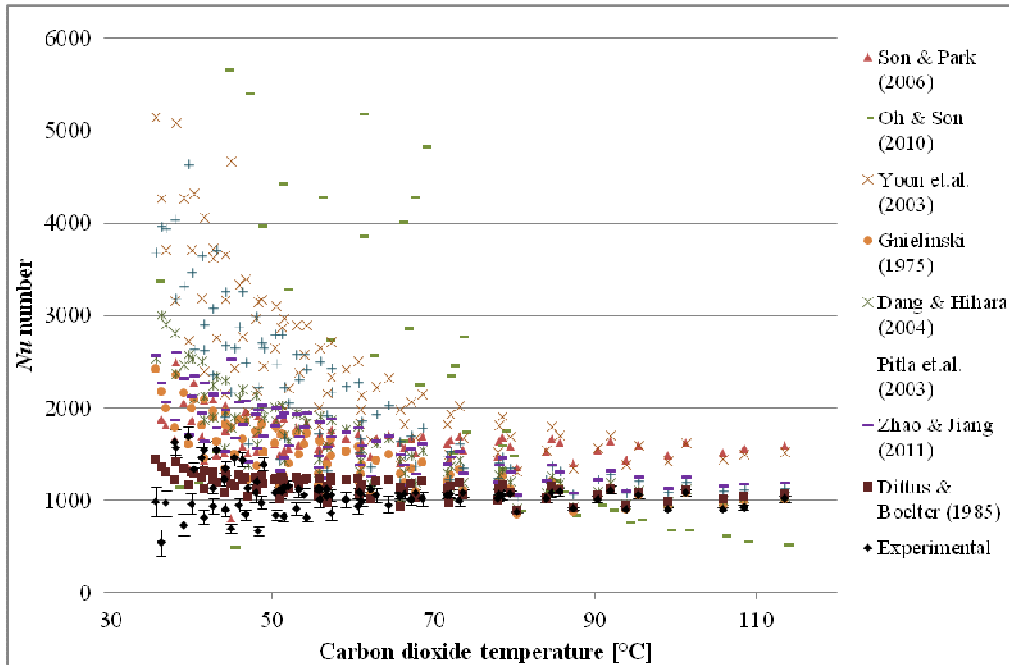


Figure 10: Nu numbers plotted against carbon dioxide temperature to illustrate the effect of fluid property variations near the pseudo-critical region.

4. CONCLUSIONS

The following conclusions were made when comparing Nu_{exp} to Nu_{theo} for the diameter tube used in the test facility:

- The Dittus and Boelter (1985) correlation predicts Nu values the best with an average relative error of 20% over the entire Re range and with an average of 9% at $350,000 < Re < 550,000$. The second best comparison was obtained by the correlation of Gnielinski (1975) with an average relative error of 45% over the entire Re range and 9.5% for Re numbers between $600,000 < Re < 700,000$.
- Nu values are generally over-predicted by most correlations.
- At temperatures close to the pseudo-critical point Nu_{theo} values responds very volatile to the change in fluid properties whereas Nu_{exp} remains relative constant. This phenomenon was observed especially in correlations where complex fluid property ratios are used.
- It is notable that the two correlations with the highest accuracies (Dittus and Boelter (1985) and Gnielinski (1975)) are the only two correlations not specifically developed for supercritical carbon dioxide cooling and therefor does not contain any property ratios as an additional factor in the correlation.
- In general the correlations evaluated in this study were developed for much lower Re number ranges than what were measured in the experimental tests. It can thus be concluded that the thermo-physical property variations plays a larger role in the Nu numbers for flow conditions with lower Re numbers, as found by numerous studies in the literature.

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NOMENCLATURE

A	Area	m^2	<i>Greek symbols</i>	
c_p	Specific heat capacity at constant pressure	J/Kg-K	ΔT	Temperature difference K
\tilde{c}_p	Integrated specific heat capacity at constant pressure	J/Kg-K	ρ	Density kg/m ³
\bar{c}_p	Mean specific heat capacity at constant pressure	J/Kg-K	<i>Subscripts</i>	
DH	Hydraulic diameter	m	b	property as measured at the bulk fluid
h	Enthalpy	J/kg	$D\&H$	Dang and Hihara (2004)
k	Conduction heat transfer coefficient	W/m-K	DB	Dittus and Boelter (1985)
L	Length	m	e	outlet
$LMTD$	Logarithmic mean temperature difference	K	exp	experimentally determined value
\dot{m}	Mass flow rate	kg/s	f	property as measured at the film layer
Nu	Nusselt number	-	G	Gnielinski (1975)
Pr	Prandtl number	-	i	inlet / inner
\dot{Q}	Heat transfer rate	W	o	outer
Re	Reynolds number	-	P	Pitla <i>et.al.</i> (2001)
T	Temperature	K	pc	property measured at the pseudo critical point
U	Overall heat transfer coefficient	W/m ²	$S\&P$	Son and Park (2006)
			$theo$	theoretically correlated value
			w	property as measured at the wall
			Y	Yoon (2003)
			$Z\&J$	Zhao and Jiang (2011)